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HEAT-TRANSFER PROCESSES IN AIR-COOLED ENGINE CYLINDERS

By Benjamin Pinkel

SUMMARY

From a consideration of heat-transfer theory, semiempirical expressions are set up for the transfer of heat from the combustion gases to the cylinder of an air-cooled engine and from the cylinder to the cooling air. Simple equations for the average head and barrel temperatures as functions of the important engine and cooling variables are obtained from these expressions. The expressions involve a few empirical constants, which may be readily determined from engine tests. Numerical values for these constants were obtained from single-cylinder engine tests for cylinders of the Pratt & Whitney 1535 and 1340-H engines. The equations provide a means of calculating the effect of the various engine and cooling variables on the cylinder temperatures and also of correlating the results of engine-cooling tests. An example is given of the application of the equations to the correlation of cooling-test data obtained in flight.

INTRODUCTION

Researches on the effect of the engine and cooling variables on the cylinder temperatures of air-cooled engines in wind tunnels and in flight and on blower-cooled single-cylinder test units, supplemented by studies of the cooling of electrically heated finned cylinders, have been made by various investigators. The results are characterized by an apparent lack of correlation. The effects of the engine and cooling variables on cylinder temperature have been found to differ for different engines and for the same engines operating at different conditions.

In the present report an analysis will be made of the heat-transfer processes of an air-cooled engine for the purpose of determining the manner in which the various engine and cooling conditions combine to determine the cylinder temperature. Besides providing a means by which the test results may be correlated, the analysis will give a better insight into the relation between the effects of the various important operating conditions and will indicate a method for reducing the testing to a minimum. The average head and barrel temperatures will each be given as a simple function of the important conditions making it possible to correct cylinder temperatures of a given engine for variation in engine power, air-fuel ratio, mass flow of cooling air, and atmospheric temperature.

In the study of the cylinder temperatures of air-cooled engines there are two processes to be considered: First, the transfer of heat from the combustion gases to the cylinder and, second, the transfer of heat from the cylinder through the fins to the cooling air. From a consideration of heat-transfer theory, semiempirical expressions for the heat transferred by these two processes will be set up. These expressions will contain constants, the values of which will be determined from tests of engine cylinders. From these expressions equations will be obtained for the average head and barrel temperatures as functions of the fundamental engine and cooling variables.

Tests were made of two modern air-cooled engine cylinders to check the analysis and to provide the necessary experimental constants.

HEAT TRANSFER FROM THE COMBUSTION GASES TO THE CYLINDER

HEAT-TRANSFER COEFFICIENT

The transfer of heat from the combustion gases to the cylinder takes place by radiation and convection. Radiation contributes to the heat transfer mainly during and immediately following the short period when the gases are burning. From tests of low-speed engines, Nusselt (reference 1) has shown that radiation accounts for less than 10 percent of the heat transferred. For high-speed engines it is reasonable to expect that radiation, as compared with convection, is of even less importance. In the present analysis the heat transfer will be assumed to take place entirely by convection.

It is now well established from theoretical considerations and tests of bodies of various shapes that, in general, the surface heat-transfer coefficients q for cooling by forced convection may be set up as functions of the Reynolds Number $\rho VD/\mu$, of the Prandtl number $\mu c_p/k$, and of nondimensional ratios of the various important dimensions of the body. Thus

$$q_g=\frac{\mu c_p}{D}f\left(\frac{\rho VD}{\mu}, \frac{\mu c_p}{k}, \frac{S}{D}, r\right) \quad (1)$$

where q_g is the rate of heat transfer per unit area per unit temperature difference between the gases of combustion and the cylinder. (The term "cylinder" is used to refer, in general, to both the head and the barrel.)
ρ, density of gas.
V, velocity of gas.
S, stroke.
D, bore.
r, compression ratio.
μ, coefficient of viscosity of gas.
c_p, specific heat of gas at constant pressure.
k, thermal conductivity of gas.

The usefulness of equation (1) is evident since, by grouping the many variables into factors such as the Reynolds Number and the Prandtl number, the testing required to establish the equation for q is reduced to a minimum.

The convection of heat from a surface by a moving gas is brought about by the same mechanism that gives rise to surface friction. It may be recalled that the friction between two adjacent fluid layers moving at different velocities is caused by an interchange of fluid particles between the layers resulting from the random molecular movement associated with fluid temperature for laminar flow and, in addition, from small swirls for turbulent flow. These same interlayer movements interchange particles in the warmer layers with particles in the colder layers and result in a diffusion of heat in the direction of the colder layers. With transition from laminar to turbulent flow, a large increase in heat transfer naturally occurs.

The values of the viscosity, conductivity, and specific heat in equation (1) vary with temperature; however, the Prandtl number for gases remains practically constant.

Heat-transfer tests have shown that, in general, q is proportional to $(\rho V)^n$ where the value of n depends on the body cooled and on the range of Reynolds Number involved. For example, in the cooling of a flat plate, $n=0.5$ for a laminar boundary layer and 0.8 for a turbulent boundary layer. It is clear that, for q_g to be proportional to $(\rho V)^n$, the function of the Reynolds Number must be $(\rho VD/\mu)^n$. The equation for q_g for a gas may then be written

$$q_g=\frac{\mu c_p}{D}\left(\frac{\rho VD}{\mu}\right)^n f\left(\frac{S}{D}, r\right) \qquad (2)$$

Several attempts have been made to obtain the variation of the instantaneous heat-transfer coefficient during the engine cycle (references 1 and 2). The difficulty of the problem is seen from the fact that, in addition to the continual variation in V, ρ, μ, c_p, and k during the cycle, account must also be taken of the varying gas turbulence. The present paper deals with the simpler problem of obtaining the average rate of heat transfer over the entire cycle.

Gas velocity.—The various gas movements that take part in the transfer of heat are the flow of the gases through the intake and exhaust ports, the sweeping of the gases over the cylinder barrel and head surfaces produced by the piston movement, and the turbulence that may be set up in the gases. A source of turbulence is the high gas velocity through the intake port, which reduces to a swirl when the gas enters the cylinder. During combustion there is evidence of the existence within the charge of a field of small swirls. As an explanation of the fact that the rate of flame propagation increased with engine speed, Marvin and his associates (reference 3) suggested that these swirls are the mechanism whereby the flame is propagated in the cylinder. It is evident that this mechanism would also transfer heat and, although little is known about it, that it may be one of the most important contributing factors. Possible sources of this small-grain turbulence may be the breaking down of the swirl introduced during the suction stroke or of an unstable flow condition set up by the rapid compression of the engine gases. Finally, additional gas turbulence may result from the friction of the gases in flowing over the cylinder surfaces and from the combustion of the charge.

In all the components of this complicated gas movement, the characteristic is evident that the linear and rotational velocities are proportional, or nearly proportional, to the engine speed. The assumption will therefore be made that the velocity V in equation (2) can be replaced by a product of the engine speed and a function of the crank angle. It will also be assumed that for a given engine the function of crank angle is independent of the engine-operating conditions.

Engine and gas parameters.—The quantities S, D, and r are constants for a given engine. Other ratios of the engine dimensions may be of importance but they also are either constants or known functions of the crank angle. The density ρ is obviously equal to the product of the weight of gases in the cylinder and a function of crank angle.

It will later be shown that the value of c_p/μ^{n-1} corresponding to an average effective gas temperature for the gas cycle does not appreciably vary for the range encountered in ordinary operation. Although c_p/μ^{n-1} varies during any given cycle, it will be assumed that its effect is the same for all cycles.

When the foregoing assumptions are introduced into equation (2), the instantaneous heat-transfer coefficient of the gas is

$$q_g=W_t^n f(\theta) \qquad (3)$$

where W_t is the product of the weight of gases in the cylinder and the number of cycles per minute, and $f(\theta)$ combines the variation with crank angle of all the factors in equation (2) and also the constants of the equation. In accordance with the preceding discussion, $f(\theta)$ is independent of the engine-operating conditions. The quantity W_t may be looked upon as the weight of gas flowing through the engine per minute adjusted to include the weight of the residuals.

Heat transfer to head.—Over the entire cycle the average of the rate of heat transfer to the cylinder head is obviously given by

$$H=\frac{a_1}{4\pi}\int_0^{4\pi} q_g(T-T_h)d\theta$$

or

$$H=\frac{a_1 W_t^n}{4\pi}\int_0^{4\pi} f(\theta)(T-T_h)d\theta \tag{4}$$

where H is the heat transferred per unit time.

a_1, the internal area of the head.

T_h, the average temperature of the head surface.

T, the instantaneous gas temperature.

It should be noted that q_g and T are average values over the cylinder head.

If B is defined as

$$B=\frac{1}{4\pi}\int_0^{4\pi} f(\theta)d\theta$$

it may be seen from equation (3) that BW_t^n is the mean heat-transfer coefficient over the complete cycle. The temperature obtained by averaging the product of the instantaneous gas temperature and the ratio of the instantaneous to the mean heat-transfer coefficient over the complete cycle will be referred to as the "effective gas temperature."

The effective gas temperature by definition is given by the following expression

$$T_g=\frac{1}{4\pi B}\int_0^{4\pi} Tf(\theta)d\theta \tag{5}$$

Equation (4) may now be written

$$H=Ba_1W_t^n(T_g-T_h) \tag{6}$$

Equation (6) thus gives a simple expression for the rate of heat transfer to the head.

The indicated horsepower I of an engine is proportional to the weight of mixture inducted into the engine per unit time for a constant air-fuel ratio. In the range to the rich side of the theoretically correct mixture, I varies only slightly with air-fuel ratio. The weight of mixture differs from the total weight in the engine by the weight of residuals. The difference, however, is sufficiently small so that an equation similar to equation (6) but involving I instead of W_t may be written

$$H=\overline{B}a_1I^{n'}(T_g-T_h) \tag{7}$$

This equation is applicable in the range of air-fuel ratios to the rich side of the theoretically correct mixture. In the lean-mixture range, I depends not only on the weight of charge inducted into the engine but also on the air-fuel ratio. For a small range to the lean side of the theoretically correct mixture, however, the effect of air-fuel ratio may approximately be taken care of by including it in the curve of T_g against air-fuel ratio. Equation (7) does not predict the small variation in heating that occurs when a constant indicated horsepower is maintained and when the weight of residuals is varied by varying the exhaust pressure.

Heat transfer to barrel.—The equations for the barrel will be assumed to take the same form as equations (5), (6), and (7). The values of the constants and of T_g will, however, be different. In order to avoid complicating the notation, the same set of symbols will be used for the barrel with the exception that T_h will be replaced by T_b, the average cylinder-barrel temperature. Part of the heat is transmitted to the barrel by the piston. The heating of the piston by the engine gases is similar in process to the heating of the cylinder head. In the conduction of heat from the piston to the barrel, however, additional factors, such as the resistance to the flow of heat through the piston, piston rings, and oil films, appear. The cooling produced by the flow of oil and the heating due to friction also affect the barrel temperatures. Some of these factors are taken care of by the empirical constants in the equations. The final justification for the foregoing assumption is, however, that good results are obtained by the use of the equations. Some variation in the constants B and $\overline{B}$ for the barrel may be expected as the condition of the piston, piston rings, and cylinder changes and also as the quantity of circulating oil changes.

Temperature of combustion gases.—The important factors upon which T_g depends can be determined from a consideration of the engine cycle. It will be shown that T_g may be taken as being independent of the weight of the charge inducted into the engine and the engine speed but depending on the air-fuel ratio, the carburetor-air temperature, the spark timing, and the compression ratio. As only first-order effects will be considered, a constant specific heat will be assumed.

Equating the heat added by combustion to the sum of the increase in internal energy and the increment of work, the familiar equation is obtained

$$dq=w_tc_vdT+\frac{Pdv}{J}$$

From the gas law

$$Pv=w_tRT$$

there is obtained $Pdv+vdP=w_tRdT$

The expression for dq then becomes

$$dq=\frac{c_v}{R}(\gamma Pdv+vdP)$$

From which

$$v^{\gamma-1}dq=\frac{c_v}{R}d(Pv^\gamma) \tag{8}$$

Integrating from the start of compression where $q=0$, $P=P_1$, and $v=v_1$ to the point up to which the quantity of heat q is liberated, there is obtained for the pressure at that point

$$P=P_1\left(\frac{v_1}{v}\right)^\gamma+(\gamma-1)\frac{J}{v^\gamma}\int_0^q v^{\gamma-1}dq$$

An expression for the average gas temperature at any instant in the cycle may be obtained from the preceding equation by making use of the gas law

$$T=\frac{P_1 v_1}{w_t R}\left(\frac{v_1}{v}\right)^{\gamma-1}+(\gamma-1)\frac{Jhw_f}{w_t v^{\gamma-1} R}\int_0^L v^{\gamma-1}dL \qquad (9)$$

where P_1 is the initial pressure of the charge when the volume is v_1.
v_1, the total volume of the cylinder with the piston at bottom center.
v, the variable volume.
γ, the ratio of the specific heat at constant pressure to the specific heat at constant volume.
J, the mechanical equivalent of heat.
q, the quantity of heat added up to any point in the cycle.
c_v, specific heat at constant volume.
L, the ratio of weight of mixture burned up to any point in the cycle to the weight inducted.
h, the heat content per pound of fuel $\times A/15.1$.
R, the gas constant.
$w_t=w+w_r$, the total weight of gas in cylinder after combustion is complete.
w_r, the weight of residuals.
$w=w_a+w_f$, the weight of fresh charge.
w_a, the weight of air.
$w_f=\frac{w}{A+1}$, the weight of fuel.
A, the air-fuel ratio.

Since the gas constant R for the residual gas is close to that for the fresh charge

$$P_1v_1=R(wT_m+w_rT_r)$$

where T_m is the temperature of the inlet charge.
T_r, the temperature of the residual gas.

$$T=\frac{(wT_m+w_rT_r)}{w_t}\left(\frac{v_1}{v}\right)^{\gamma-1}+\frac{(\gamma-1)hwJ}{R(A+1)w_t}\left(\frac{v_1}{v}\right)^{\gamma-1}\int_0^L\left(\frac{v}{v_1}\right)^{\gamma-1}dL$$

The number of variables in this equation may be reduced by eliminating the ones having only a small effect on T. The quantity w_rT_r is equal to P_dv_c/R, where P_d is the pressure at the end of the exhaust stroke and v_c is the clearance volume of the cylinder. The quantity P_d would be expected to increase slightly with weight of charge. The term w_rT_r, however, is only a small factor in determining the value of T and, in seeking a first approximation, the effect of variation in the quantity w_rT_r/w_t will be neglected. When w is large compared with w_r, w_t is practically proportional to w, and the variation of w/w_t with weight of charge is small. For the purpose at hand it will be assumed that w/w_t is constant. This assumption applies fairly well at full open throttle and its accuracy decreases as the load is decreased. The quantity L is a function of crank angle and, for a given air-fuel ratio, spark timing, and cylinder, has been found to vary only slightly with weight of charge and engine speed. Curves illustrating this point will be shown later.

The expression for T may then be written

$$T=T_m\left(\frac{v_1}{v}\right)^{\gamma-1}+f_1(\theta, r, s, A)$$

where r is the compression ratio.
s, the spark timing.
θ, the crank angle at which T is the gas temperature.

This equation applies during the compression and expansion strokes.

During exhaust

$$T=T_m+f_1(2\pi, r, s, A)$$

and, during intake, neglecting the residuals,

$$T=T_m \text{ (approximately)}$$

It is evident that an expression of the form

$$T=T_mf_2(r, \theta)+f_3(\theta, r, s, A) \qquad (10)$$

covers the variation of T over the entire cycle. It is noted that, based on the foregoing assumptions, T is independent of the quantity of charge, the weight of residuals, and the engine speed in the practical range of operation.

When equation (10) is substituted into equation (5), there results

$$T_g=T_mf_4(r)+f_5(r, s, A) \qquad (11)$$

The value of T_g is also independent of the charge inducted, the weight of residuals, and the engine speed, since $f(\theta)$ in equation (5) is independent of the engine conditions. The variation of T_g for both the head and barrel with A, s, and T_m will be obtained from engine tests. The values of the constants B, $\overline{B}$, n, and n' for the head and the barrel will also be obtained from engine tests.

HEAT TRANSFER FROM CYLINDER TO COOLING AIR

Over-all heat-transfer coefficient.—The transfer of heat from the cylinder wall through the fins to the cooling air will now be considered. It is possible to obtain theoretical solutions for the rate of flow of heat through fins as a function of the fin dimensions and the surface heat-transfer coefficient of the fins. Harper and Brown (reference 4) have made a detailed analysis of the heat flow through fins of rectangular, tapered, and parabolic section mounted on flat plates and cylinders. The following convenient approximate expression given in reference 5 has been found to predict with a good degree of accuracy the heat transfer for circular fins of both rectangular and tapered section:

$$U=\frac{q}{p}\left[\frac{2}{a}\left(1+\frac{w}{2R_b}\right)\tanh(aw')+s_b\right] \qquad (12)$$

where

$$a=\sqrt{\frac{2q}{kt}}$$

U is the rate of heat dissipation per unit cylinder-wall area per degree difference in temperature between the cylinder wall and the air.

q, the average surface heat-transfer coefficient of fins is the rate of heat dissipation per unit surface area of the fins per degree difference in temperature between the fin surface and the air.

w, width of fin.

p, fin pitch $(s+t)$.

s, average air space between fins.

t, average fin thickness.

w', width of fin plus ½ tip thickness.

s_b, length of cylinder wall exposed between two adjacent fins.

R_b, radius to base of fins.

k, thermal conductivity of fin material.

Equation (12) may be written

$$U=bq^x$$

where b and x are functions of the fin dimensions and of the value of q. For a given engine the fin dimensions are fixed and the variation of q is limited to the range corresponding to adequate cooling. Within this range b and x are found to be constants for a given engine.

The surface of the fin is cooled by convection and an equation similar to equation (1) can be written for the heat-transfer coefficient q where D is replaced by s, and r, by s/R_b. As indicated in the previous discussion, q is proportional to $(\rho V)^n$. In this case also the variation of the viscosity μ with cooling-air temperature introduces only a small variation in q, which will be neglected.

Since the pressure drop $\Delta P\rho/\rho_0$ across a finned cylinder mounted in a jacket or in a cowling varies as a power of the mass flow, the following expression may be written for U:

$$U=K(\Delta P\rho/\rho_0)^m \tag{13}$$

where K and m are constants for a given engine and ρ_0 is a standard density (taken as corresponding to 70° F. at 29.92 inches of Hg). The quantity ρ is the average density of the air entering and leaving the fins. The values of K and m will differ for the head and barrel.

Heat transfer to cooling air.—The heat transferred from the head to the cooling air is given by

$$H=a_0K(\Delta P\rho/\rho_0)^m(T_h-T_a) \tag{14}$$

where a_0 is the outside head area.

T_h, the average head temperature.

T_a, the inlet temperature of the cooling air.

A similar expression may be obtained for the barrel.

Average temperatures.—If H in equation (14) is equated to the value given in equation (7), an expression can be set up for the average head temperature:

$$T_h-T_a=\frac{T_g-T_a}{\frac{a_0K(\Delta P\rho/\rho_0)^m}{a_1BI^{n'}}+1} \tag{15}$$

An expression similar to (15) but involving a different set of constants may be written for the barrel. The average head and barrel temperatures, by means of these expressions, can be calculated from the engine and cooling conditions.

An equation similar to (15), but involving the weight of gas in the engine cylinder, may be obtained from equations (14) and (6):

$$T_h-T_a=\frac{T_g-T_a}{\frac{a_0K(\Delta P\rho/\rho_0)^m}{a_1BW_t^n}+1} \tag{16}$$

Maximum cylinder temperature.—The rear spark-plug temperature and the temperature between the rear spark plug and the exhaust port are the highest external temperatures on the head. The practice at present is to set a temperature of about 475° F. as the maximum that a point on the cylinder head may attain. A maximum of 300° F. is also usually imposed on barrel temperatures.

No attempt will be made here to obtain an equation for the temperature at individual points on the cylinder as a function of the cooling and engine variables. The variation of the average head and barrel temperatures may, however, be taken as a close indication of the variation of the maximum cylinder temperatures when engine and cooling conditions are changed. The maximum temperatures have been found to exceed the average temperatures by from 120° F. to 150° F. on the head and 30° F. on the barrel for the Pratt & Whitney 1340-H cylinder in the single-cylinder tests. Different values depending on the fin and baffle design will be obtained for other cylinders.

The rate of heat transfer from the hot gases to the cylinder wall varies from point to point on the cylinder. In the case of the barrel it may be assumed that the heat-transfer coefficient from the gases to the wall is the same for any point at a given height on the barrel but that it varies as the point moves axially. The problem is even more complicated for the head, as the position of the point relative to the spark plugs results in a difference in heating during combustion. In addition, various parts of the head are subject to different degrees of gas movement. The region near the intake port is cooled by the flow of the fresh charge into the cylinder, and the region near the exhaust port is heated by the movement of hot gases at high velocity through the exhaust port.

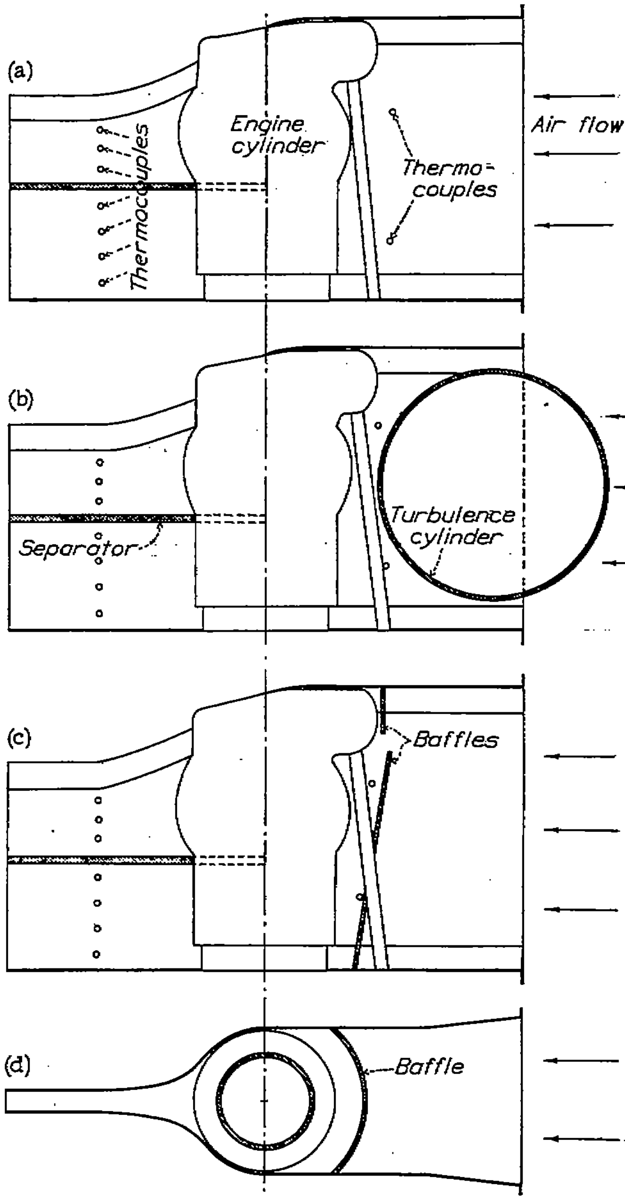
The rate of heat transfer from the cylinder wall to the cooling air also varies with position on the cylinder. The cooling of the front of the cylinder depends on the degree of turbulence of the air. Inferior cooling of the rear of the cylinder results from the heating of the air as it flows around the cylinder and from the poor flow conditions directly at the rear.

In the treatment of the cooling of individual points on the cylinder, consideration must also be given to the conduction through the metal from the hot regions to the cold, and particularly in the case of the cylinder head because of the high conductivity of aluminum and the thickness of the walls.

It is evident that, even if it could be obtained, an equation taking all these factors into consideration would be complicated and of little practical value.

APPARATUS

The cylinders on which the tests were made were a Pratt & Whitney 1535 and a Pratt & Whitney 1340–H.



(a) Engine cylinder and jacket.
(b) Engine cylinder, jacket, and turbulence cylinder.
(c) Engine cylinder, jacket, and baffles.
(d) Section plan view of cylinder, jacket, and baffles.

FIGURE 1.—Diagram of engine cylinder, jacket, and turbulence devices.

The bore, stroke, and compression ratio for the 1535 cylinder were 5$\frac{3}{16}$ inches, 5½ inches, and 6.73, respectively, and for the 1340–H were 5¾ inches, 6 inches, and 5.6. Each cylinder was mounted on a single-cylinder test stand enclosed in a jacket (see fig. 1 (a)) through which cooling air was forced by a centrifugal blower The jacket had a wide entrance section to provide a low air velocity over the front half of the cylinder; over the rear half the jacket fitted closely against the fins to provide a high air velocity.

The 1535 cylinder was provided with two devices for disturbing the air flow over the front of the cylinder to determine the additional cooling due to the disturbed motion of the air in front of the cylinders of a cowled engine in flight. One of the devices was a cylinder 9 inches in diameter mounted about 4 inches ahead of the fin tips with its axis at right angles to the cylinder axis, as shown in figure 1(b). The other device consisted of baffles for directing the cooling air downward over the front of the head and barrel, as shown in figures 1(c) and (d). The lower baffle, a semicircular sheet placed about 1½ inches from the fin tips, provided a fairly high vertical velocity of the air.

A partition was located in the exit duct for separating the air that flowed over the head from the air that flowed over the barrel. Thermocouples were located in the jacket ahead of the cylinder and in the exit passages from the head and barrel for measuring the increase in temperature of the cooling air. The jacket of the 1340–H cylinder was covered with felt insulation to reduce heat losses from the cooling-air stream.

The temperatures of the two cylinders were measured in each case by 22 thermocouples on the cylinder head, 10 on the barrel, and 2 on the flange, located in similar positions for both cylinders.

A static tube was located in the space ahead of the cylinder, where the velocity head was negligible, for measuring the pressure difference between the front of the cylinder and the room. Impact tubes were located between the fins on the head and barrel at 45° from the front of the cylinder. The space between the fins and the jacket was still large in the region of the impact tubes. As the velocity head was only a small part of the total head in this region, there was little loss in energy and the readings of the impact tubes were the same as that of the static tube. In the tests with the turbulence devices the pressure drop was obtained from the readings of the impact tubes.

The quantity of cooling air supplied to the jacket was determined from the readings of a thin-plate orifice tank connected to the inlet of the blower. Electrical heaters located in the air duct between the blower and the jacket were provided to vary the cooling-air temperature. Electrical heaters were also located in the intake system of the engine for varying the carburetor-air temperature.

The standard test-engine equipment was used for measuring brake mean effective pressure, engine speed, fuel consumption, and air temperature at the inlet to the carburetor. The mixture strength was determined by means of a Cambridge air-fuel ratio meter and was used for calculating the weight of carburetor air supplied to the engine. A gasometer was connected to the intake manifold for determining the air quantity supplied to the engine during the tests in which the air-fuel ratio was varied. These measurements were used for calibrating the Cambridge meter. A Farnboro indicator was used in the tests of the 1340–H cylinder to obtain indicator cards.

METHODS AND TESTS

The following tests were made of both cylinders:

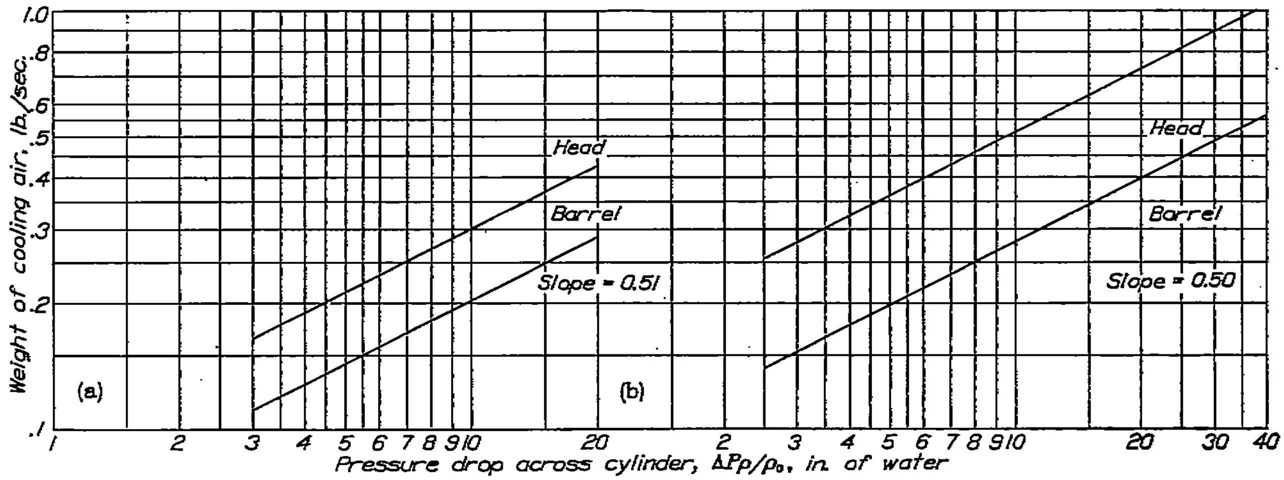
1. Calibration tests were made to determine the weight of air flowing over the head and the barrel as functions of the pressure drop across the cylinder.
2. The engine conditions and weight velocity of the cooling air were maintained constant and the cooling-air temperature was varied to determine the value of T_g at an air-fuel ratio of 12.5.
3. The cooling conditions and air-fuel ratio were held constant (the latter at a value of about 12.5) and the indicated horsepower was varied by varying the brake mean effective pressure and engine speed to determine the constants B, $\overline{B}$, n, and n' in equations (6) and (7).

The tests covered the following range of conditions:

	Pratt & Whitney 1535	Pratt & Whitney 1340–H
Engine speed, r. p. m.	1,300–2,100	1,500–2,100
Brake mean effective pressure, lb./sq. in.	72–126	75–117
Air-fuel ratio	11.3–14.7	10.6–14.1
Cooling-air temperature, °F	100–216	80–215
Pressure drop across cylinder, in. water	5–15	7.8–25
Carburetor-air temperature, °F	[1] 80	88–215
Spark timing, deg. B. T. C.	[1] 20	17–32

[1] Approximately.

More tests than were necessary to establish the values of the constants were made to check the validity of the equations over a range of engine and cooling conditions. Only two accurate tests are required to establish the value of the constants for the normal operating range of an engine: The pressure drop across the cylinder is varied in one and the indicated horsepower is varied in the other. From the first test, the constants of equation (14) may be obtained; and, from the second, the constants of equations (6) and (7).



(a) The 1535 cylinder. (b) The 1340-H cylinder.

FIGURE 2.—Calibration of the jacket as an air duct.

4. The engine conditions were held constant and the mass flow of the cooling air was varied to determine the constants m and K in equation (14). These tests provided additional data for determining T_g.
5. The cooling conditions and the weight of air delivered to the engine per cycle were held constant and the air-fuel ratio was varied to determine the variation of T_g with air-fuel ratio.

In addition, the following tests were made of the 1340–H cylinder:

6. The weight of the charge and the air-fuel ratio were held constant and the spark setting was varied to obtain the variation of T_g with spark timing.
7. The weight of the charge and the air-fuel ratio were held constant and the temperature of the carburetor air was varied to obtain the variation of T_g with carburetor-air temperature.
8. Indicator cards were obtained in runs during which the brake mean effective pressure and engine speed were varied to determine their effect on L.

The calibration tests (item 1) were made by blocking, in turn, the exits for the head and the barrel and by measuring with the thin-plate orifice tank the quantity of air flowing through the jacket for various pressure drops across the cylinder. A correction, which was obtained by blocking both passages, was applied for the leakage from the jacket. The calibration curves are shown in figure 2.

The heat dissipated from the head and barrel was calculated by the formula

$$H=W_c c_p \Delta T$$

where W_c is the weight of air flowing over the head or barrel per unit time as obtained from figure 2, and ΔT is the increase in cooling-air temperature.

The values of T_g for the head and barrel, corresponding to an air-fuel ratio of 12.5, were obtained from the tests described in item 2 by plotting the heat output from the head and the barrel against the average head and barrel temperatures, respectively. The curves were extrapolated to zero heat output, at which points the average head and barrel temperatures are equal to their respective effective gas temperatures T_g. (See equation (7).) The average head and barrel temperatures are an average of the 22 thermocouple readings for the head and of the 10 for the barrel, respectively.

The values of $H/(T_g-T_h)$ and $H/(T_g-T_b)$ were plotted against I and W_t on logarithmic coordinates for the tests described in item 3 to obtain the constants B, $\overline{B}$, n, and n' in equations (6) and (7). The values of T_g found in the manner described in the preceding paragraphs were used in calculating $H/(T_g-T_h)$ and $H/(T_g-T_b)$, inasmuch as the air-fuel ratios were approximately 12.5.

These constants and equations (6) and (7) were used to calculate the values of T_g from the tests described in items 5, 6, and 7, and the results were plotted against air-fuel ratio, spark setting, and carburetor-air temperature, respectively.

By the use of the data obtained in the tests described in item 4, the heat-transfer coefficients U were calculated for the head from $H/[a_0\ (T_h-T_a)]$ (see equation (14)) and for the barrel from a similar equation. The values of K and m were obtained from a plot on logarithmic coordinates of U against $\Delta P\rho/\rho_0$.

The curves for L were calculated from the indicator cards, making use of a convenient equation derived from equation (8).

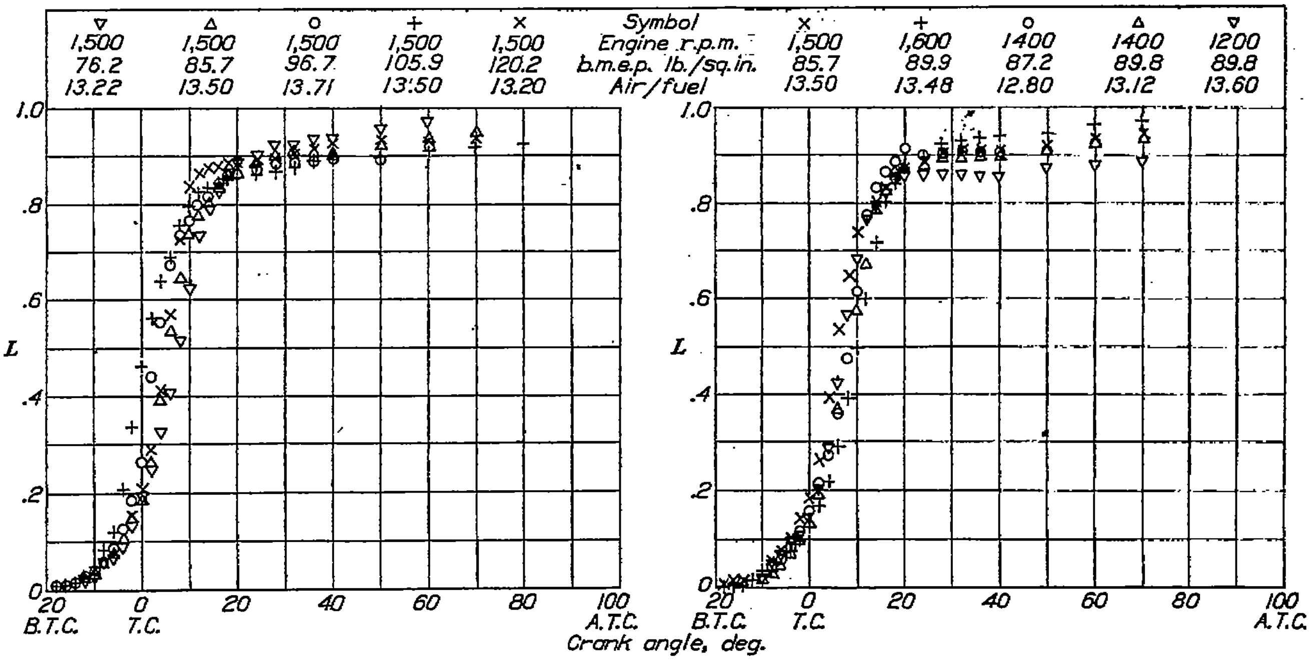


FIGURE 3.—Values of L for Pratt & Whitney 1340-H cylinder for different engine conditions.

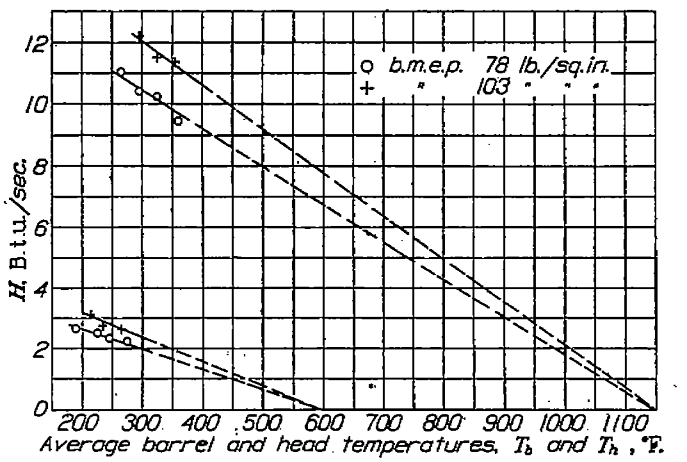


FIGURE 4.—Variation of heat transfer from combustion gases to head and barrel with average head and barrel temperatures. Carburetor-air temperature, 85°F.; spark timing, 26° B. T. C.; air-fuel ratio, 12.3; engine speed, 1,500 r. p. m.

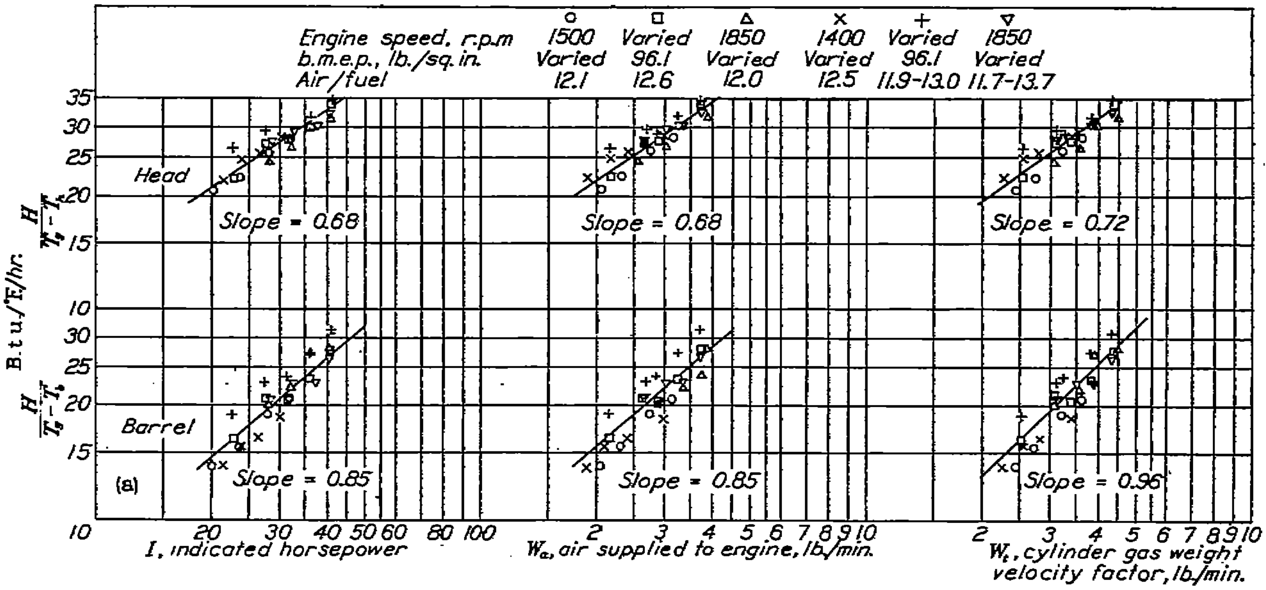
RESULTS AND DISCUSSION

Gas temperature T_g.—The values of L are shown plotted in figure 3 against crank angle for various engine brake mean effective pressures and engine speeds. As pointed out in the development of the expression for T_g, there is only a small spread between the values of L for these different conditions.

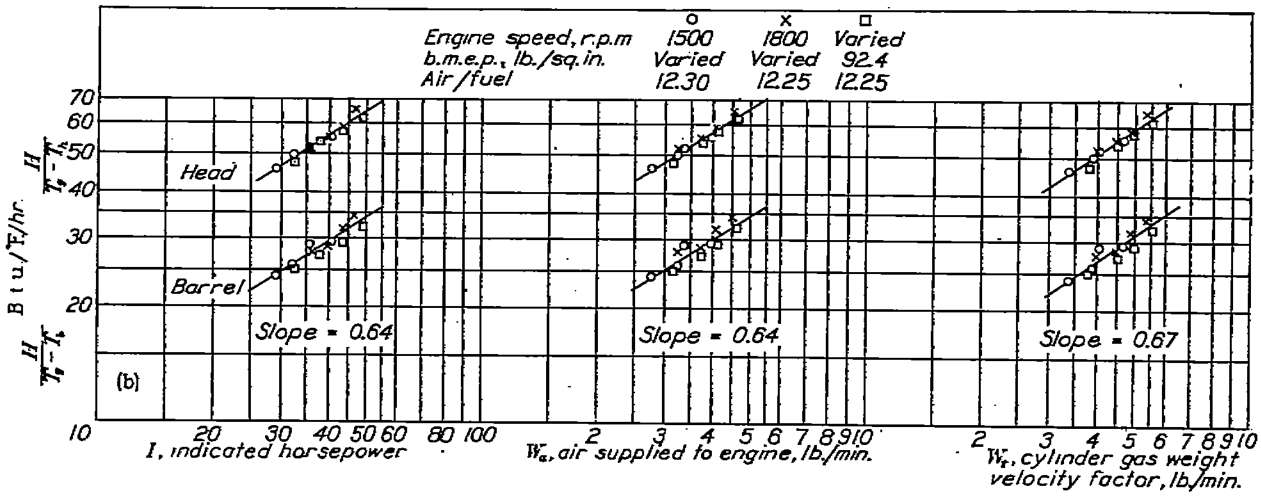
Some scattering was found in the values of the gas temperature T_g, obtained as described in the section on methods, by extrapolating the curves of heat output from the head and the barrel plotted against the average head and barrel temperatures. Part of the scattering was due to the large distance through which the extrapolation had to be made as compared with the range covered by test data and part was due to small experimental errors in the data. Consideration of a large number of these curves led to a choice of 1,150° F. and of 600° F. for the values of T_g for the head and the barrel, respectively, for both the 1535 and 1340-H cylinders. Figure 4 shows two typical sets of data for determining T_g for the Pratt & Whitney 1340-H cylinder. The lines are drawn through the points to pass through the temperatures of 1,150° F. and 600° F. at $H=0$ for the head and barrel, respectively. The curves illustrate that a single determination cannot, in general, be relied on to determine T_g.

Heat transfer from combustion gas to cylinder.—The values of $H/(T_g-T_h)$ and $H/(T_g-T_b)$ were calculated using the preceding values of T_g; they are plotted in figure 5 on logarithmic coordinates against I, W_a, and W_t for both cylinders. The points for the head of the 1535 cylinder fall on the same curve whether the change in indicated horsepower was obtained by varying the brake mean effective pressure or the engine speed, as was expected from a consideration of equations (6) and (7). The same general result was found for the head and barrel of the 1340–H cylinder. The points for the barrel of the 1535 cylinder show more scattering, although the points for the various runs form straight lines having a common slope. A line was drawn through the average of the points and in the direction indicated by the individual runs. The parallel shifting from one run to the next does not indicate a trend with brake mean effective pressure or engine speed because the results of the tests in which the brake mean effective pressure was varied and the engine speed was held constant are parallel to those for which the reverse was true. Check runs taken on different days also show about the same order of magnitude of shift. The scattering is attributed to the poor condition of the piston rings of the 1535 cylinder and the results are considered as not very reliable. Other tests of a Pratt & Whitney 1535 engine indicate that the value of n' for the barrel is the same as the value for the head.

In two of the runs plotted in figure 5 the air-fuel ratio varied over a small region in the rich range. The points for these two runs agree with the rest of the data. It will later be seen that, for the range covered, the variation of T_g with air-fuel ratio is small.



(a) The 1535 cylinder.



(b) The 1340-H cylinder.

FIGURE 5.—Variation of $H/(T_g-T_h)$ and $H/(T_g-T_b)$ with I, W_a, and W_t.

The curves of $H/(T_g-T_h)$ and $H/(T_g-T_b)$ plotted against I and W_a are parallel, having a slope of 0.68 for the head and of 0.85 for the barrel for the 1535 cylinder and of 0.64 for both the head and the barrel for the 1340–H. This parallelism is to be expected, as the indicated horsepower is directly proportional to the air weight per second. The slopes of the curves of $H/(T_g-T_h)$ and $H/(T_g-T_b)$ plotted against W_t are 0.72 for the head and 0.96 for the barrel of the 1535 cylinder and 0.67 for both the head and the barrel of the 1340–H.

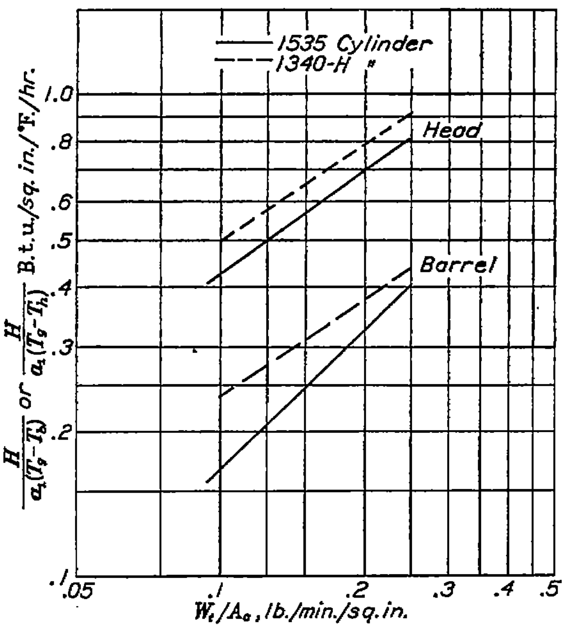


FIGURE 6.—Variation of heat-transfer coefficient from combustion gases to head and barrel with W_t/A_c.

The constants for equations (6) and (7) were obtained from figure 5; the equations are tabulated as follows:

	Head	Barrel	
1535	$H=2.71\ I^{0.68}\ (T_g-T_h)$ $H=13.5\ W_a^{0.68}\ (T_g-T_h)$ $H=11.6\ W_t^{0.72}\ (T_g-T_h)$	$H=1.185\ I^{0.85}\ (T_g-T_b)$ $H=8.46\ W_a^{0.85}\ (T_g-T_b)$ $H=6.68\ W_t^{0.96}\ (T_g-T_b)$	(17)
1340-H	$H=5.22\ I^{0.64}\ (T_g-T_h)$ $H=23.8\ W_a^{0.64}\ (T_g-T_h)$ $H=20.0\ W_t^{0.67}\ (T_g-T_h)$	$H=2.77\ I^{0.64}\ (T_g-T_b)$ $H=12.4\ W_a^{0.64}\ (T_g-T_b)$ $H=10.3\ W_t^{0.67}\ (T_g-T_b)$	

Units of H are B. t. u. per hour; units of W_a and W_t are pounds per minute.

A comparison will be made between the two cylinders of the rate of heat transfer from the gas to the cylinder walls. Dividing $H/(T_g-T_h)$ by the internal area of the head and doing the same for the barrel gives the heat-transfer coefficient of the gas to the head and the barrel. The internal area of the head included the exposed internal surface of the head with the valves closed and the surface of the liner down to the end of the head. The area of the valve faces was included but the area of the exhaust passage above the valve was not. Although the exhaust-passage area should be considered, it is difficult to state its importance in comparison with the internal area of the head. The coefficients, as previously calculated, may be considered as approximate values for purposes of comparison. The internal area of the barrel was taken from the end of the head to the position of the bottom compression ring when the piston was at bottom center.

In order to compare the heat-transfer coefficients of the two cylinders, the ratio of W_t to the cross-sectional area of the piston was taken as a measure of the weight velocity of the gas moving by the cylinder surfaces.

Curves of $H/[a_1(T_g-T_h)]$ and $H/[a_1(T_g-T_b)]$ are shown in figure 6 plotted against W_t/A_c, where A_c is the internal cross-sectional area of the cylinder. The values of a_1 and A_c are given in the following table:

	a_1, sq. in.		A_c, sq. in.
	Head	Barrel	
Pratt & Whitney 1535	47.1	82.6	21.1
Pratt & Whitney 1340–H	76.8	84.2	25.9

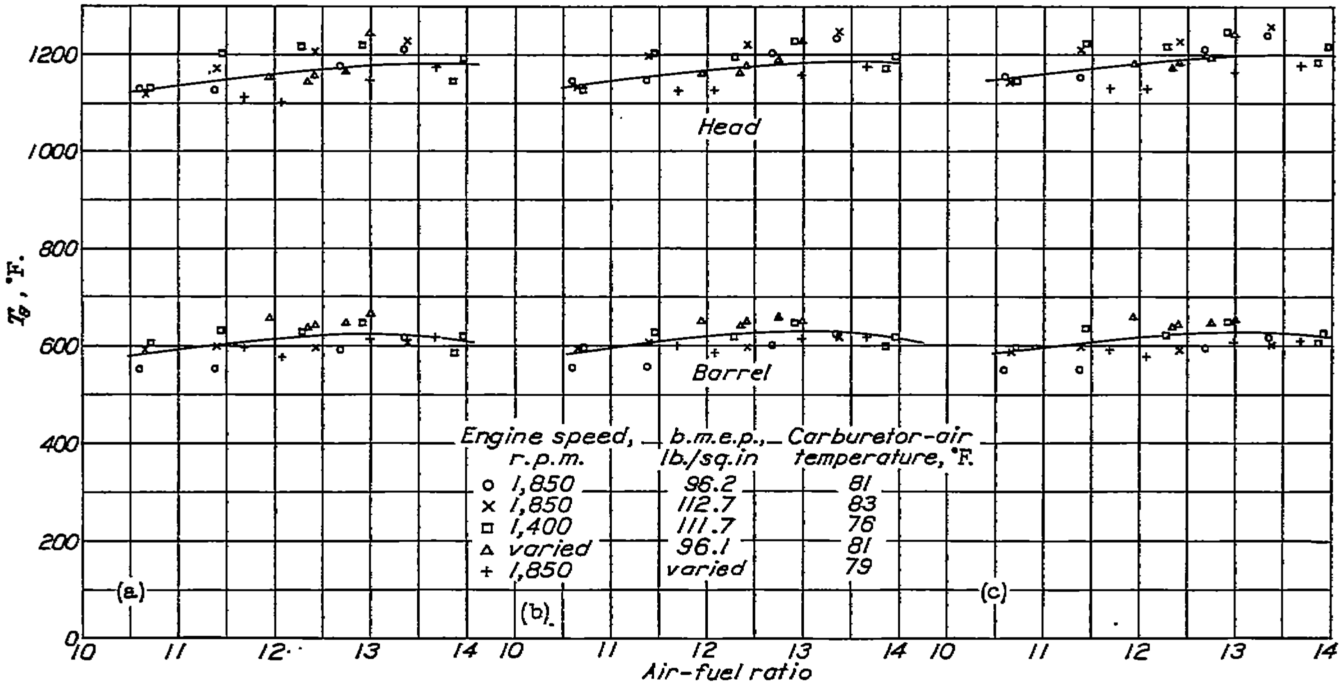
The curves for the 1340–H cylinder are somewhat higher than those for the 1535. It was expected that the results for the two cylinders would be fairly close together as there is no great difference in their general design. A large difference would be expected if means were provided in one of the cylinders for obtaining additional swirl of the charge. Much closer agreement between the two cylinders is obtained when $H/[a_1(T_g-T_h)]$ and $H/[a_1(T_g-T_b)]$ are plotted against W_t.

Variation of T_g with engine conditions.—Equations (17) were used to calculate T_g from the test results in which air-fuel ratio and spark timing were varied. Three curves were obtained for the head and barrel by using in turn the equations involving I, W_a, and W_t. Figure 7 shows the value of T_g plotted against air-fuel ratio. The three curves for T_g for each cylinder are practically the same and indicate that T_g increases only slightly in going from an air-fuel ratio of 12 to 14.

Figure 8 shows the values of T_g plotted against spark setting for the 1340–H cylinder. The value of T_g remains practically constant over a wide range of spark setting covering usual operating conditions; it increases, however, for both greatly advanced and retarded spark timings. Although W_a was held practically constant, the indicated horsepower decreases at both extremes of the spark-timing range.

In figure 9 T_g is shown plotted against the temperature of the carburetor air for the 1340–H cylinder. It was found, in general, to be difficult to obtain consistent runs in the tests of the effect of carburetor-air temperature on cylinder temperature. The curves in figure 9 indicate that a 100° F. rise in carburetor-air temperature results in approximately a 58° F. rise in T_g for the head and barrel.

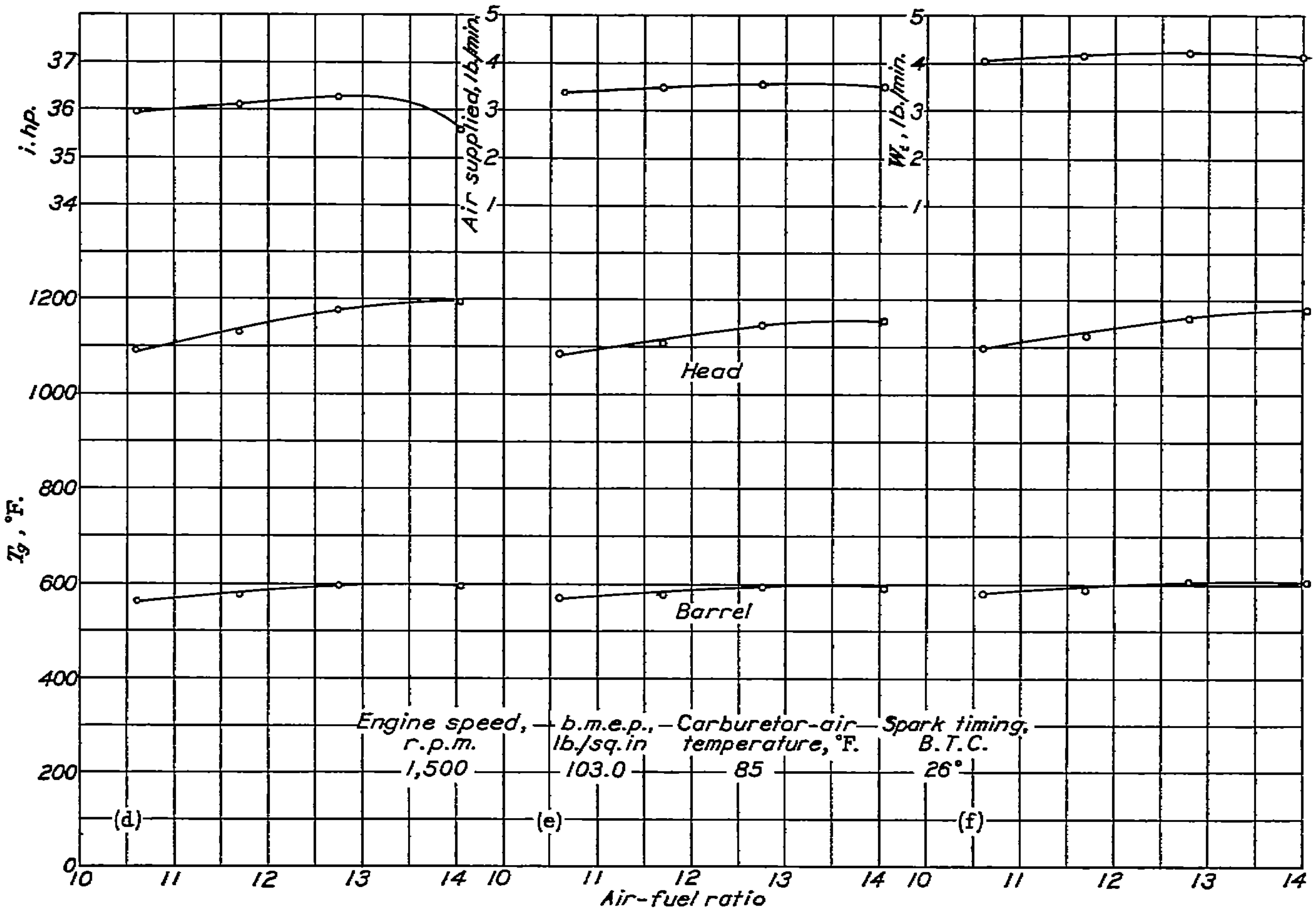
If reference is had to equations (2) and (3), an estimate may be made of the error introduced by assuming that the value of $\mu^{1-n}c_p$ corresponding to T_g does not vary appreciably with change in engine conditions. The value of n for the head is about 0.7.



(a) T_g based on indicated horsepower, I; 1535 cylinder.

(b) T_g based on air supplied to engine, W_a; 1535 cylinder.

(c) T_g based on cylinder gas weight flow factor, W_t; 1535 cylinder.

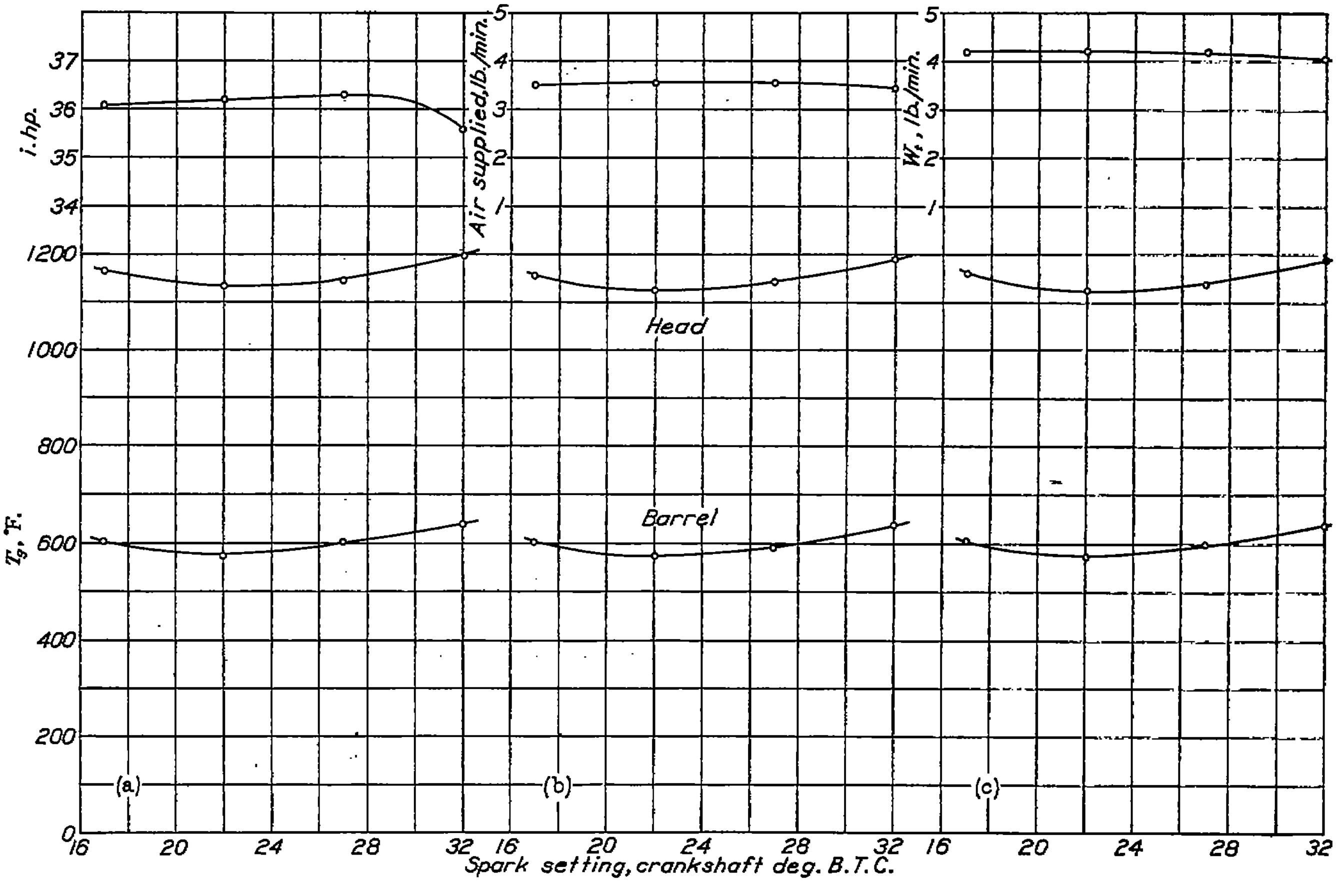


(d) T_g based on indicated horsepower, I; 1340-H cylinder.

(e) T_g based on air supplied to engine, W_a; 1340-H cylinder.

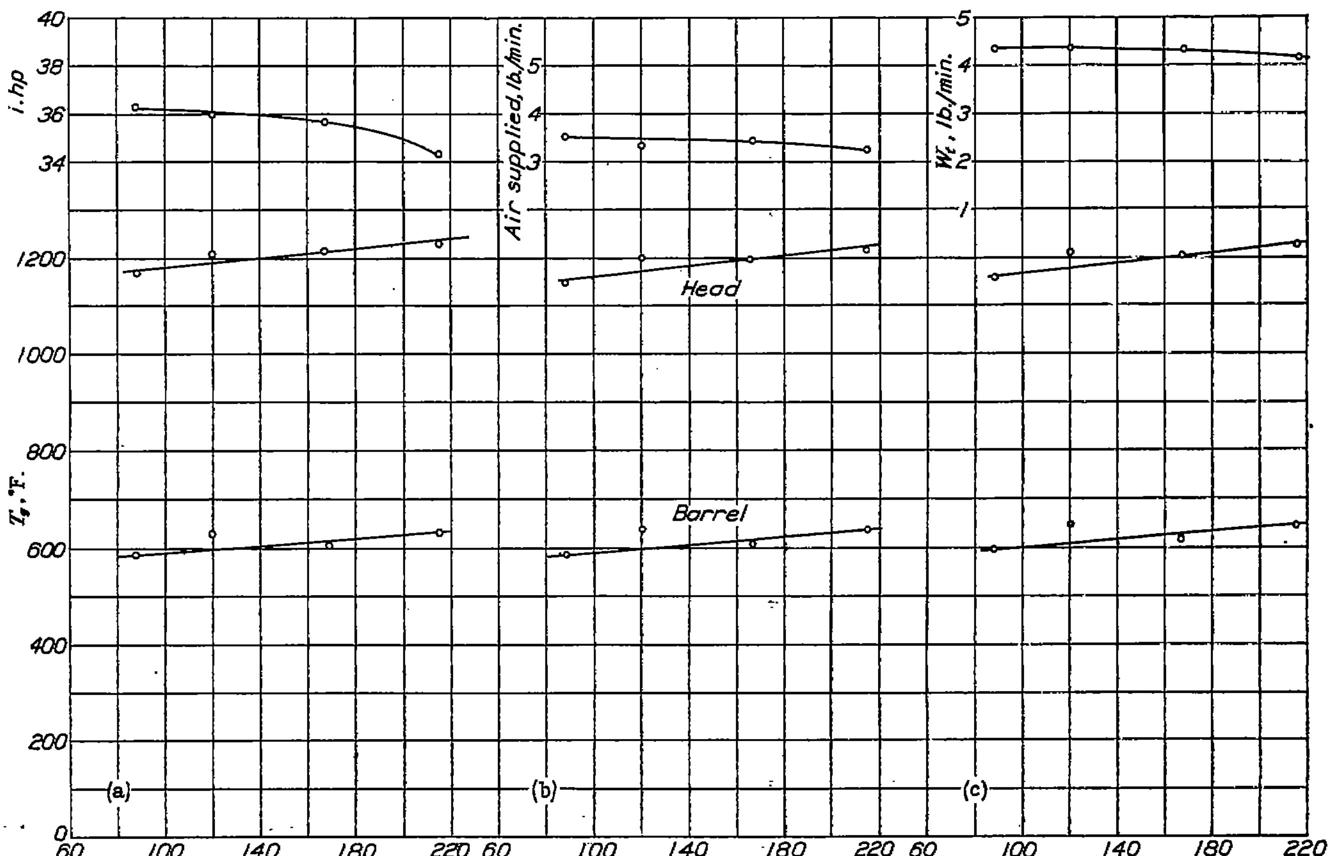
(f) T_g based on cylinder gas weight flow factor, W_t; 1340-H cylinder.

FIGURE 7.—Variation of T_g with air-fuel ratio.



(a) T_g based on indicated horsepower, I. (b) T_g based on air supplied to engine, W_a. (c) T_g based on cylinder gas weight flow factor, W_t.

FIGURE 8.—Variation of T_g with spark setting. The 1340-H cylinder; engine speed, 1,500 r. p. m.; b. m. e. p., 102.0 pounds per square inch; air-fuel ratio, 12.34; carburetor-air temperature, 93° F.



(a) T_g based on indicated horsepower, I. (b) T_g based on air supplied to engine, W_a. (c) T_g based on cylinder gas weight flow factor, W_t.

FIGURE 9.—Variation of T_g with carburetor-air temperature. The 1340-H cylinder; engine speed, 1,500 r. p. m.; b. m. e. p., 101.5 pounds per square inch; air-fuel ratio, 12.25; spark timing, 26° B. T. C.

The heat-transfer coefficient q must, therefore, vary as $\mu^{0.3}c_p$. Assuming that the effective gas temperature T_g is 1,150° F., a change in gas temperature of 100° F. will produce a change in μ of 4.8 percent and a negligible change in c_p resulting in a change in the value of q of 1.6 percent. As the variation of T_g is less than 100° F. for the usual range of variation of air-fuel ratio, carburetor-air temperature, and spark timing, it is evident that the variation in q resulting from the change in μ and c_p may be neglected.

Heat transfer from cylinder to cooling air.—The heat-transfer coefficients U for the 1535 cylinder are shown in figure 10(a) plotted against $\Delta P\rho/\rho_0$ for three conditions: Without a turbulence device, with the 9-inch cylinder, and with the vertical baffles. The 9-inch cylinder and the baffles illustrate the effect of providing turbulence over the front of the cylinder, a condition which is present in the cooling of an engine in flight. To reproduce flight conditions on a single-cylinder blower-cooled engine involves several natural difficulties. The nature of the turbulence produced by any device of the type described is different from that obtained in flight and, in addition, part of the air that cools the front of the cylinder in flight spills out of the cowling, resulting in a lower temperature of the air that enters the baffles and cools the rear of the cylinder. With regard to turbulence, it may be sufficient, without exactly reproducing the flow, to provide a large-scale turbulence in which is dissipated the same percentage of the available dynamic head as in flight. The present results may be considered to indicate the effect of providing turbulence over the front of the cylinder.

The curves for the three conditions are parallel and have a slope of 0.35 for the head and 0.31 for the barrel. Similar curves for the 1340–H cylinder without any turbulence device are shown in figure 10(b). The slopes of these curves are 0.34 for the head and barrel. Plotted in figure 10 for comparison with the over-all heat-transfer coefficient U of the barrel fins are curves calculated by means of equation (12) using the fin dimensions of the barrel and values of q obtained from blower cooling tests on electrically heated, finned cylinders (reference 6).

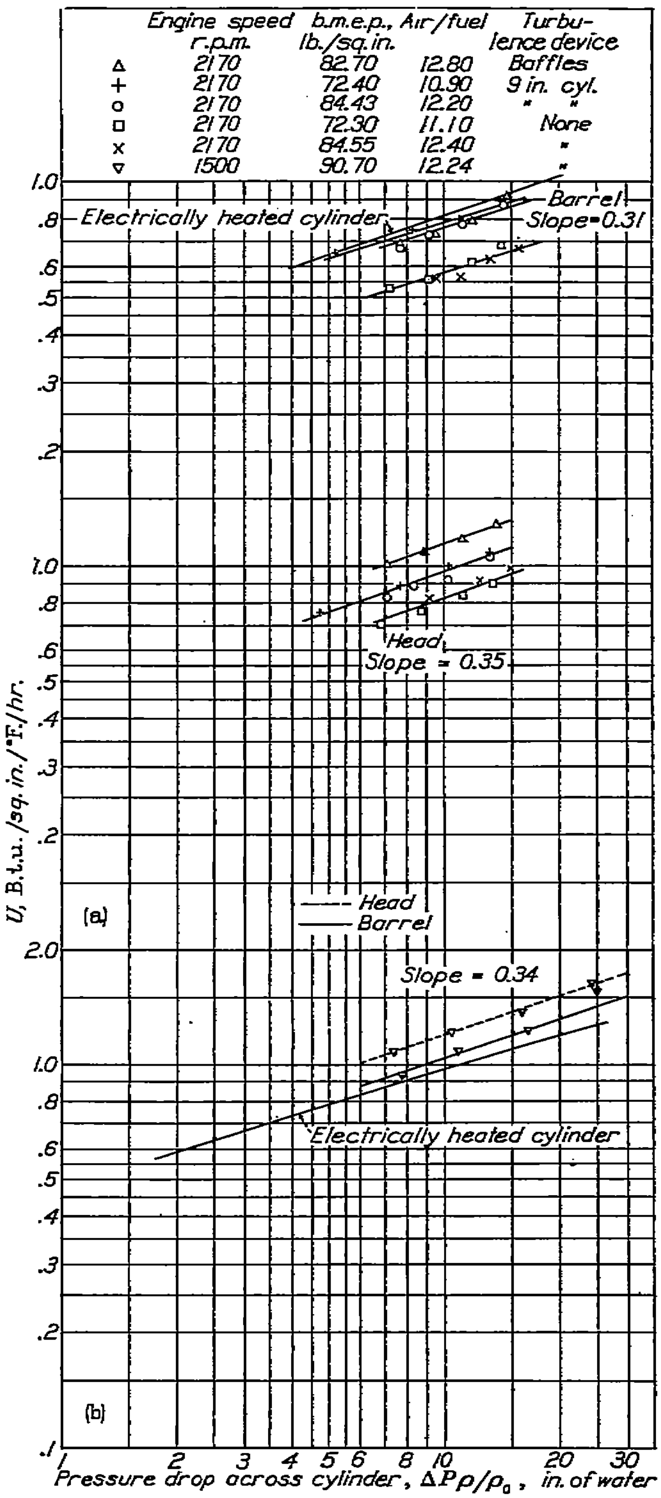
The equations for the heat transferred to the cooling air obtained from figure 10 are tabulated as follows:

Cylinder	Turbulence device	Head	Barrel
1535	None	$H=34.5\ (\Delta P\rho/\rho_0)^{0.35}(T_h-T_a)$.	$H=17.1\ (\Delta P\rho/\rho_0)^{0.31}(T_b-T_a)$.
1535	9-inch cylinder.	$H=40.5\ (\Delta P\rho/\rho_0)^{0.35}(T_h-T_a)$.	$H=22.9\ (\Delta P\rho/\rho_0)^{0.31}(T_b-T_a)$.
1535	Baffles	$H=47.6\ (\Delta P\rho/\rho_0)^{0.35}(T_h-T_a)$.	$H=22.3\ (\Delta P\rho/\rho_0)^{0.31}(T_b-T_a)$.
1340–H	None	$H=78.1\ (\Delta P\rho/\rho_0)^{0.34}(T_h-T_a)$.	$H=33.0\ (\Delta P\rho/\rho_0)^{0.34}(T_b-T_a)$.

(18)

Units of H are B. t. u./hr.; units of ΔP are inches of water.

For the 1535 cylinder the area a_0 for the outside barrel surface covered by fins is 61.5 square inches and for the head is 94.5 square inches. The areas are 68.5



(a) The 1535 cylinder. (b) The 1340–H cylinder.

FIGURE 10.—Variation of heat-transfer coefficient of fins on head and barrel with pressure drop across cylinder.

and 142 square inches, respectively, for the barrel and head of the 1340–H cylinder.

The heat-transfer coefficients obtained when the 9-inch cylinder and the baffles were used are, respectively, 17 and 38 percent higher on the head and 34 and 30 percent higher on the barrel than for the condition of no turbulence.

Average head and barrel temperatures.—If H is eliminated between equations (17) and (18), expressions may be obtained for T_h and T_b in the form given by equations (15) and (16). For example, if the expression for the heat transfer as a function of the indicated horsepower is selected from equations (17) and the expression for the heat transfer corresponding to the case of no turbulence device from equations (18), the following expressions give the average temperature of the head and barrel, respectively, for the 1535 and 1340–H cylinders.

$$\left.\begin{array}{l}\text{Pratt \& Whitney 1535 cylinder:}\\ \qquad\text{Head}\\ T_h-T_a=\dfrac{T_g-T_a}{12.7\dfrac{(\Delta P\rho/\rho_0)^{0.35}}{I^{0.68}}+1}\\ \qquad\text{Barrel}\\ T_b-T_a=\dfrac{T_g-T_a}{15.1\dfrac{(\Delta P\rho/\rho_0)^{0.31}}{I^{0.85}}+1}\\ \text{Pratt \& Whitney 1340–H cylinder:}\\ \qquad\text{Head}\\ T_h-T_a=\dfrac{T_g-T_a}{15.0\dfrac{(\Delta P\rho/\rho_0)^{0.34}}{I^{0.64}}+1}\\ \qquad\text{Barrel}\\ T_b-T_a=\dfrac{T_g-T_a}{11.9\dfrac{(\Delta P\rho/\rho_0)^{0.34}}{I^{0.64}}+1}\end{array}\right\}\quad(19)$$

An examination of figures 8 and 9 indicates that a value for T_g of 1,150° F. for the head and of 600° F. for the barrel holds fairly well for the range of air-fuel ratios and spark settings used in practical operation.

Figure 11 shows the values of T_h and T_b for the 1340–H cylinder, calculated by means of equation (19) using the foregoing values of T_g, compared with values obtained from tests in which the indicated horsepower, ΔP, and T_a were varied independently. A fair agreement will be noted.

Expressions similar to (19) for the average head and barrel temperatures of the 1535 cylinder, but applying more closely to flight conditions, may be obtained by using an equation in set (18) corresponding to one of the turbulence devices. For instance, by the use of the equations for the baffles, the following expressions will be obtained:

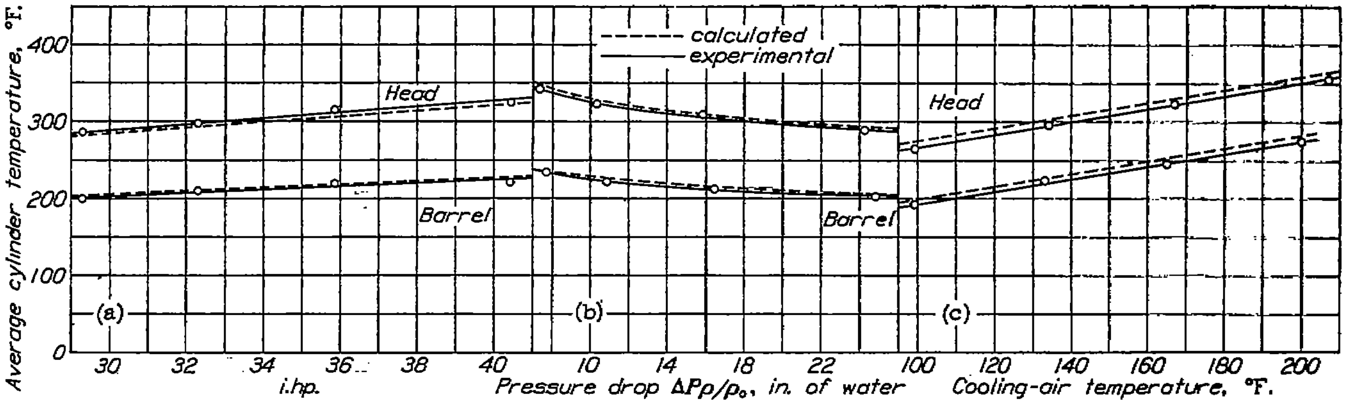
$$\left.\begin{array}{l}T_h-T_a=\dfrac{T_g-T_a}{17.55\dfrac{(\Delta P\rho/\rho_0)^{0.35}}{I^{0.68}}+1}\\ T_b-T_a=\dfrac{T_g-T_a}{19.55\dfrac{(\Delta P\rho/\rho_0)^{0.31}}{I^{0.85}}+1}\end{array}\right\}\quad(20)$$

The effect of providing turbulence was to increase the value of the coefficient of $(\Delta P\rho/\rho_0)^m/I^{n'}$. Where considerable change in the weight of residuals is experienced as in flight at various altitudes, an expression involving W_t instead of I is more accurate (i. e., as given by equation (16)).

Equations (19) and (20) indicate that, to maintain a constant average head temperature, the pressure drop $\Delta P\rho/\rho_0$ must vary approximately as the square of the indicated horsepower or, since the mass flow varies almost as the square root of the pressure drop, the mass flow must be increased approximately in direct proportion to the indicated horsepower.

The heat-transfer coefficient U for the head is proportional to $(\Delta P\rho/\rho_0)^m$ and it is evident from equations (19) and (20) that, as U is increased (by improving the fin design or the air flow), the indicated horsepower may be increased approximately according to the relation

$$\frac{I_2}{I_1}=\left(\frac{U_2}{U_1}\right)^{1.5}$$



(a) Variation with indicated horsepower; air-fuel ratio, 12.3; pressure drop, 28 inches of water.

(b) Variation with pressure drop; air-fuel ratio, 12.25; b. m. e. p., 90.7 pounds per square inch.

(c) Variation with cooling-air temperature; air-fuel ratio, 12.25; b. m. e. p., 78.4 pounds per square inch; pressure drop, 25 inches of water.

FIGURE 11.—Comparison of calculated and experimental values of average head and barrel temperatures at various test conditions. The 1340-H cylinder; engine speed, 1,500 r. p. m.

APPLICATIONS

1. Equations (15) and (16) provide a means for plotting the results of all cooling tests for a given engine on a single curve. For example, equation (15) may be written

$$\frac{T_g-T_h}{T_h-T_a}I^{n'}=\left(\Delta P\frac{\rho}{\rho_0}\right)^m\times\text{constant}$$

or

$$\frac{T_g-T_h}{T_h-T_a}I^{n'}=K\left(V\frac{\rho}{\rho_0}\right)^\gamma$$

where γ is approximately equal to $2m$ as $\Delta P\rho/\rho_0$ varies as a power of the mass flow in the neighborhood of 2. power. Take 115 miles per hour as the velocity of climb at sea level. For a constant indicated air speed the velocity in climb at 30,000 feet is

$$V_{alt}=V_{SL}\sqrt{\frac{\rho_{SL}}{\rho_{alt}}}=115\sqrt{\frac{1}{0.374}}=188 \text{ m. p. h.}$$

At sea level, $T_a=59°$F., $\frac{\rho_{SL}}{\rho_0}=\frac{530}{519}=1.02$

since ρ_0 is taken in this report as the density at a temperature of 70° F. and standard sea-level pressure.

$$V\frac{\rho}{\rho_0}=1.02\times115=117 \text{ m. p. h.}$$

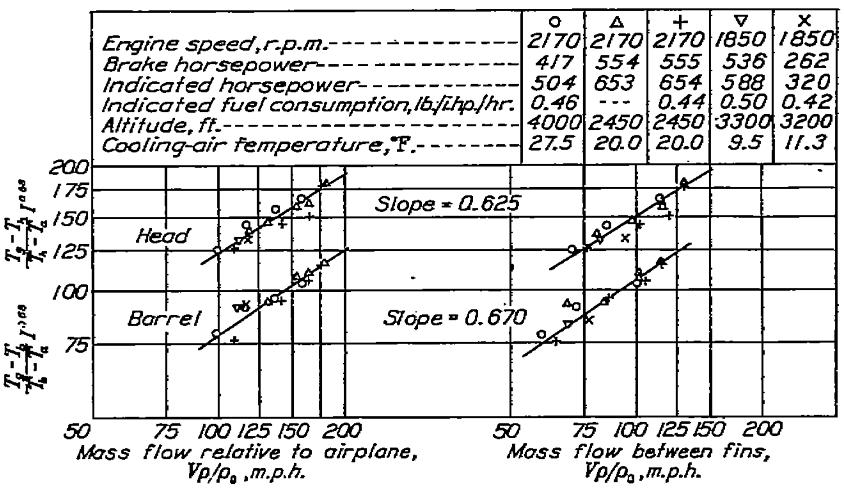


FIGURE 12.—Variation of $(T_g-T_h/T_h-T_a)I^{0.68}$ and $(T_g-T_b/T_b-T_a)I^{0.68}$ with mass flow from flight tests of a cylinder of a Pratt & Whitney 1535 engine in a Grumman Scout airplane.

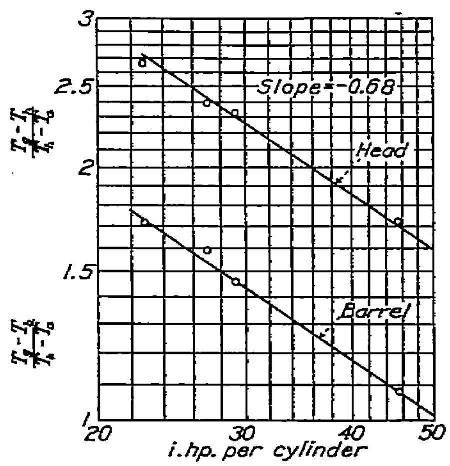


FIGURE 13.—Variation of T_g-T_h/T_h-T_a and T_g-T_b/T_b-T_a with indicated horsepower for a constant mass flow of cooling air from flight tests of a cylinder of a Pratt & Whitney 1535 engine in a Grumman Scout airplane. $V\rho/\rho_0=134$ m. p. h. (relative to airplane).

Figure 12 shows $(T_g-T_h/T_h-T_a)\ I^{n'}$ obtained in flight plotted against $V\rho/\rho_0$ for a cylinder of a Pratt & Whitney 1535 two-row radial engine mounted in a Grumman Scout airplane. The details of the tests are given in reference 7. The cooling-air velocity between the fins was found to be proportional to the airplane velocity for the range covered in the tests. The values of T_g corresponding to the air-fuel ratios obtained in the tests were taken from figure 6. A value for n' of 0.68 for the head and for the barrel was obtained in the flight tests. (See fig. 13.) This value confirms that obtained for the head in the laboratory tests of the Pratt & Whitney 1535 cylinder and was used in plotting figure 12. It will be noted that the test points in figure 12 fall on a single curve, irrespective of engine conditions. The slopes of the curves differ slightly from the slopes that might be expected from the laboratory tests of the 1535 cylinder.

2. Assume that the Grumman airplane is equipped with a supercharger capable of maintaining an indicated horsepower of 550 up to an altitude of 30,000 feet. A comparison is desired of the cylinder-head temperatures in climb at sea level and at 30,000 feet in a standard atmosphere at the same indicated air speed and engine power.

At 30,000 feet $\frac{\rho_{alt}}{\rho_0}=0.374\times\frac{530}{519}=0.381$

$$V\frac{\rho}{\rho_0}=188\times0.381=71.6 \text{ m. p. h.}$$

$$I=550 \text{ hp. } I^{0.68}=73$$

The carburetor-air temperature and air-fuel ratio are assumed constant and T_g is taken as 1,150° F.

From figure 12

at sea level $T_a=59°$ F. $\quad\frac{T_g-T_h}{T_h-T_a}I^{0.68}=135$

$$\frac{1150-T_h}{T_h-59}=\frac{135}{73}=1.85$$

$$T_h=441° \text{ F.}$$

at 30,000 feet $T_a=-48°$ F. $\quad\frac{T_g-T_h}{T_h-T_a}I^{0.68}=99$

$$\frac{1150-T_h}{T_h+48}=\frac{99}{73}=1.355$$

$$T_h=460° \text{ F.}$$

An increase in average head temperature of only 19° F. is indicated. The preceding calculation neglects the

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reduction in the weight of residual gas in the cylinder with increase in altitude. If the variation in the weight of the residual gas were considered, the temperature increase would be even less than 19° F.

The cylinder temperatures were assumed to have attained equilibrium in both cases. Because of the heat capacity of the cylinder materials, the cylinder temperatures do not respond instantly to a change in engine or cooling conditions. This fact tends to complicate the comparison of cylinder temperatures obtained in climb tests. Where large changes in conditions occur, as when going from level flight into a climb, a fast-climbing airplane may climb to considerable height before the cylinder temperatures are within 10° F. of equilibrium.

CONCLUSIONS

1. The heat-transfer coefficient for the transfer of heat from the combustion gases to the cylinder head varied as the 0.68 power of the indicated horsepower for the Pratt & Whitney 1535 cylinder and as the 0.64 power of the indicated horsepower for the Pratt & Whitney 1340-H cylinder.

2. The heat-transfer coefficient for the transfer of heat from the combustion gases to the cylinder barrel varied as the 0.68 power of the indicated horsepower for the Pratt & Whitney 1340-H cylinder.

3. The values of the effective gas temperatures were practically independent of the engine speed and brake mean effective pressure and for the normal range of operation were equal to 1,150° F. for the head and 600° F. for the barrel for both the Pratt & Whitney 1535 and the 1340-H cylinders.

4. The values of the effective gas temperature (a) decreased slightly as the air-fuel ratio decreased on the rich side of the theoretically correct mixture, (b) remained constant for a range of spark timing to either side of normal operation and increased for both extremely retarded and advanced spark timing, and (c) increased 58° F. for 100° F. increase in the inlet-air temperature for the Pratt & Whitney 1340-H cylinder.

5. The rate of heat transfer from the head and the barrel to the cooling air varied, respectively, as the 0.35 and 0.31 powers of the pressure drop across the cylinder for the Pratt & Whitney 1535 cylinder and as the 0.34 power for both the head and the barrel of the 1340-H cylinder.

6. The turbulence devices in front of the cylinder provided an increase in the heat-transfer coefficient of the order of 30 percent for the same pressure drop.

7. For the cylinders tested, in order to maintain a constant cylinder-head temperature for a given cylinder, it was necessary to increase the pressure drop across the cylinder directly as the square of the indicated horsepower.

8. The equations indicate that to improve the heat transfer from the fins by improving the fin design or by using higher air speeds allows the indicated horsepower to be increased as the 1.5 power of the heat-transfer coefficient U with practically no increase in cylinder-head temperature.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., *June 11, 1937.*

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